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EFFECTIVENESS OF PNEUMATIC CONVEYING SYSTEMS FOR COOLING SPRAY-DRIED FOOD PRODUCTS

SUMMARY—The cooling effectiveness during pneumatic transport of a spray-dried food product has been investigated. Differential equations which describe the product and air temperatures as a function of distance from the initial mixing point have been derived and solved. The predicted results have been compared to experimental data obtained in a conveying tube equipped with thermocouples. The results indicated the equilibrium temperature of the air-product mixture could be predicted and was a function of loading ratio. Increased loading ratio decreased the distance required to reach the equilibrium temperature. The effectiveness of product cooling after reaching the equilibrium temperature was a function of conditions at the conveying tube wall. Results indicated that a water spray over the exterior surface was more effective than forced-air circulation which, in turn, was more effective than natural air circulation.

INTRODUCTION

THE TYPE of cooling system used for cooling of spray-dried food products will vary considerably depending on facilities available. In most cases, the cooling is accomplished by mixing the dry product with room temperature or cool air followed by conveying through a tube to the separators. Use of room-temperature air and long conveying tubes is normally adequate to reduce the product temperature to acceptable levels unless the ambient temperatures are too high. This situation does occur during the summer season in many parts of the United States and very frequently in countries with warm climates.

The dry product cooling problem is further complicated by the concern for providing purified air as a conveying medium. Recent encounters with *Salmonellae* contamination have emphasized the need for more effective control over product-air contact. Purified air can be provided by high-efficiency filters. However, large volumes are required as a conveying medium and power requirements become rather large. Through supplemental cooling methods, the power requirements for conveying and cooling of the dry food products may be kept to a minimum.

Objectives of this investigation were to a) investigate the mechanisms of product cooling during initial mixing as product and air temperature come to equilibrium and b) determine the effectiveness of supplemental cooling of the conveying tube wall on product cooling. The first objective was accomplished by performing an enthalpy balance on a section of conveying tube and solving the equations for air and product temperature as a function of distance. Preliminary verification was accomplished by conducting experiments in an insulated conveying tube. At locations beyond the equilibration region, the influence of forced-air movement and water spray over exterior portions of the conveying tube has been investigated.

There is an obvious lack of information in the literature on the effectiveness of cooling methods for powdered materials, although fluidized systems have been investigated to a considerable extent (Depew and Farbar, 1963; Soo, 1967; Soo et al. 1960; Tien, 1961; Wilkinson and Norman, 1967). In addition, there seems to be no published data which indicate the optimum storage temperatures for dry food products. Hanrahan et al. (1967) have made reference to slow cooling resulting in a cooked or scorched flavor in spray-dried milk, and indicated that sinkability of dry milk decreased significantly when the product was stored at 20°C rather than cooled to 5°C. The only logical conclusion is that maximum quality is attained only by effective cooling before the product reaches the storage container. Results by Farrall et al. (1968) and Chen (1969) indicate that thermal conductivities of dry food powders are sufficiently low to prevent rapid cooling of the product in the storage container.

Theoretical considerations

Cooling of dry food particles by cool air during pneumatic flow occurs in 2 steps a) a region immediately after the product is released into the air stream during which the product and air come to some equilibrium temperature and b) the region after the equilibration, during which the product and air cool by heat transfer to the walls of the conveying tube.

The following analysis will deal with the equilibration region, assuming adiabatic conditions exist at the conveying tube walls. By derivation of equations which describe both the enthalpy content of the air and the product as it moves through the conveying tube (Fig. 1), the change in temperature of product and air can be predicted, based on properties of product and air.

Symbols

A	=	particle surface area per unit volume of product-air mixture
C	=	specific heat
D	=	diameter
h	=	convective heat transfer coefficient
k	=	thermal conductivity
L.R.	=	loading ratio, lb product per lb air
q	=	heat flux
S	=	cross-sectional area of tube
T	=	cooling medium temperature
V	=	velocity
W	=	weigh density of powder per unit volume; Equation [19].
x	=	axial distance in tube
Pr	=	Prandtl number $= \frac{C_a \mu_a}{k_a}$
Nu	=	Nusselt number $= \frac{h_p D_p}{k_a}$

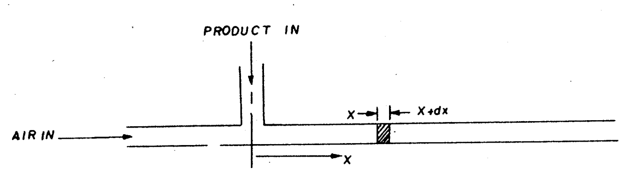

Fig. 1—Schematic diagram.

Re	=	Reynolds number for tube = $\frac{\rho_a D V_a}{\mu_a}$
$(Re)_d$	=	Modified particle Reynolds number = $\frac{V_a D_p}{\nu}$
Re_p	=	particle Reynolds number = $\frac{\rho_a D_p (V_a - V_p)}{\mu_a}$
St	=	Stanton number defined in Equation [6]
B_1	=	dimensionless number defined in Equation [6]
B_2	=	dimensionless number defined in Equation [7].
θ	=	product temperature
ρ	=	density
μ	=	viscosity

Subscripts

a	=	air
p	=	product
o	=	initial conditions
w	=	wall conditions

Enthalpy balance on air

Using a control volume within the conveying tube as a reference, the enthalpy difference between x and x + dx would be:

$$V_a S \rho_a C_a (T + \frac{dT}{dx}) dx - V_a S \rho_a C_a T = V_a S \rho_a C_a \frac{dT}{dx} dx \quad [1]$$

Since the process being considered is adiabatic, heat transfer will occur only between the particles and air as described by the following equation:

$$q = - h_p A S (T - \theta) dx \quad [2]$$

The right-hand sides of Equations [1] and [2] can be equated to obtain the following differential equation:

$$\frac{dT}{dx} + \frac{h_p A}{V_a \rho_a C_a} (T - \theta) = 0 \quad [3]$$

which describes the temperature of the air during the equilibration region.

Enthalpy balance on products

Using an approach similar to the enthalpy balance on air, the enthalpy of the product will change some amount as it passes through the control volume as described by the following expression:

$$V_p \rho_p C_p S (\theta + \frac{d\theta}{dx}) dx - V_p \rho_p C_p S \theta = V_p \rho_p C_p S \frac{d\theta}{dx} dx \quad [4]$$

By equating the right-hand sides of Equations [2] and [4], the differential equation describing product particle temperature is obtained:

$$\frac{d\theta}{dx} = \frac{h_p A}{V_p \rho_p C_p} (T - \theta) \quad [5]$$

Solutions for differential equations

Equations [3] and [5] can be solved by methods normally used for solving ordinary differential equations.

By letting:

$$B_1 = \frac{h_p A}{V_a \rho_a C_a} = \frac{\frac{h_p D_p}{k_a} A}{\frac{V_a D_p}{\nu} \cdot \frac{C_a \mu}{k_a}} = \frac{(Nu)_d A}{(Re)_d Pr} = (St)_d A \quad [6]$$

$$B_2 = \frac{h_p A}{V_p \rho_p C_p} = \frac{\frac{h_p D_p}{k_a} A}{\frac{V_a D_p}{\nu} \cdot \frac{C_a \mu}{k_a} \cdot \frac{V_p \rho_p C_p}{V_a \rho_a C_p}} = \frac{(Nu)_d A}{(Re)_d Pr W} = \frac{(St)_d A}{W} \quad [7]$$

then: $$\frac{dT}{dx} + B_1 (T - \theta) = 0 \quad [8]$$

$$\frac{d\theta}{dx} = B_2 (T - \theta) \quad [9]$$

or: $$T = \frac{1}{B_2} \frac{d\theta}{dx} + \theta \quad [10]$$

By substitution of Equation [10] into [8], a new differential equation is obtained:

$$\frac{1}{B_2} \frac{d^2\theta}{dx^2} + (1 + \frac{B_1}{B_2}) \frac{d\theta}{dx} = 0 \quad [11]$$

Equation [11] can be solved to obtain:

$$\theta = \frac{1}{1 + w} \left\{ (w \theta_o + T_o) + (T_o - \theta_o) \exp \left[- B_1 (1 + \frac{1}{w}) x \right] \right\} \quad [12]$$

where:

$$w = \frac{B_1}{B_2} \quad [13]$$

while:

$$T = \frac{1}{1 + w} \left\{ (w \theta_o + T_o) + w (T_o - \theta_o) \exp [- B_1 (1 + \frac{1}{w}) x] \right\} \quad [14]$$

Equations [12] and [14] can be written in dimensionless form as follows:

$$\frac{\theta - \theta_o}{T_o - \theta_o} = \frac{1}{1 + w} \left\{ 1 - \exp [-(St)_d (1 + \frac{1}{w}) Ax] \right\} \quad [15]$$

and:

$$\frac{T - T_o}{\theta_o - T_o} = \frac{w}{1 + w} \left\{ 1 - \exp [- (St)_d (1 + \frac{1}{w}) Ax] \right\} \quad [16]$$

Application of derived equations

Equations [15] and [16] allow prediction of the air and product temperatures as a function of distance during pneumatic flow from the heat transfer coefficient between particle and air (h_p), particle surface area per unit volume (A), velocities of air and particles (V_a, V_p), density of air and particles (ρ_a, ρ_p) and specific heats of air and particles (C_a, C_p). Probably the most difficult of these parameters to measure is the heat transfer coefficients between particle and air. An empirical expression has been reported by Rowe et al. (1965) as follows:

$$Nu = 2 + b (Re_p)^a (Pr)^{0.33} \quad [17]$$

where:

$b = 0.459$ for air

$a = 0.4$ at $Re_p = 1$ and 0.6 at $Re_p = 10^4$

To use Equation [17] the particle Reynolds number (Re_p) must be computed which, in turn, is dependent on relative velocity between particle and air. The Hinkel equation for computing this relative velocity was reported by Zenz (1960):

$$\frac{V_p}{V_a} = 1 - 0.179 D_p^{0.3} \rho_p^{0.5} \quad [18]$$

Assumptions and limitations

The assumptions in the above derivation are as follows: The temperature range is sufficiently narrow that air and product properties do not vary significantly. The temperature differences are small enough to make radiation heat transfer insignificant. Temperature gradients do not exist within particles due to small size, and mass transfer does not occur due to low-moisture contents.

One potential limitation would be failure to consider heat transfer between particles and conveying tube walls. However, Farbar and Morley (1957) indicated this contribution should be very small due to small contact area between particle and wall, and short contact time. Probably the more significant limitations are related to predicting the heat transfer coefficient between particle and air. Use of Equation [18] for predicting relative velocity between particle and air assumes that the value is constant throughout the equilibration region. Actually, the particle may be accelerating in velocity during a considerable portion of the equilibration region. In addition, Equation [17] assumes the particles are well dispersed, which is not the situation during the early portions of the temperature equilibration between product and air.

EXPERIMENTAL

TWO TYPES of experiments were conducted. The region of flow which includes the equilibration of product and air temperature has been investigated under adiabatic conditions. The region of cooling after product and air have equilibrated has been investigated under three different surface-cooling conditions.

Experimental apparatus

The equipment used is shown schematically in Figure 2. It consisted of a section of 2-in.-

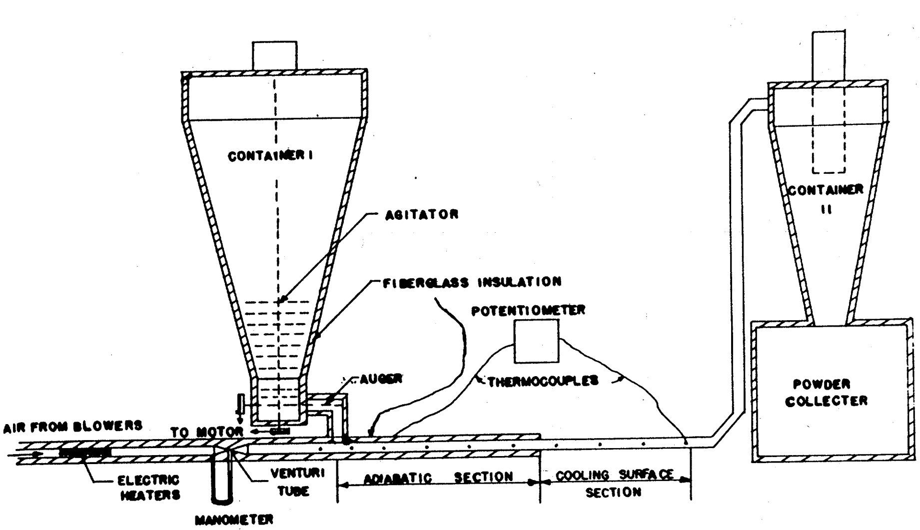

Fig. 2—Diagram of experimental apparatus.

diameter tubing into which a dry food powder could be metered and conveyed pneumatically to a powder collector.

A stainless steel powder collector (I) was used as a source of product. An agitator which turned around a vertical axis and a 2-in.-diameter auger were added to the container. Air was drawn from the room at flow rates measured with a venturitube and U-tube manometer.

Immediately downstream from the location of product injection, an 8-ft section of 2-in. (id) plexiglass tubing was insulated to provide the adiabatic conditions for product-air equilibration. The section of tubing used for testing the cooling effectiveness was connected to the adiabatic section and consisted of a 5-ft section of 2-in. (id) aluminum tubing. Temperature measurements in the tube were made at either 6-in. or 1-ft intervals, depending on location, using copper constantan thermocouples and 12-point recording potentiometer. The temperature sensors, located 0.75-in. from the bottom surface of the tube, indicated the mean temperature of the 2-phase flow. In addition, thermocouples were located in the product container and upstream from the site of product injection to provide initial product and air temperatures, respectively.

Product characteristics

Nonfat dry milk was used in all experiments. The mean particle diameter was assumed to be 60 μ, based on results reported by Hayashi et al. (1968) for products manufactured by the same procedures as in this investigation. A particle density of 1.46 was used, based on data by Hall and Hedrick (1966). A specific heat of 0.36 Btu/lb-°F was predicted, based on product composition and verified by experimental determination in a Differential Thermal Analyzer. This heat transfer surface area was computed by a method proposed by Zenz (1960) as:

$$A = \frac{6W}{\rho_p\ D_p} \qquad [19]$$

where W is the weight density per unit volume of fluidized power in air.

Test procedures

A typical experiment involved an initial heating of all powder to between 115 and 140°F. During experiments dealing with mixing of product and air resulting in equilibration of temperature in an adiabatic section of tubing, loading ratios (L.R.) of 0.54, 0.78 and 1.17 were utilized.

The effectiveness of various surface cooling methods was investigated in the tubing immediately downstream from the equilibration section. These tests were conducted using a loading ratio of 0.73 and three different surface-cooling methods: a) natural air circulation, b) forced-air circulation and c) water spray. The procedure involved the measurement of the air-product mixture temperatures at 0.5-ft intervals throughout the test section.

RESULTS & DISCUSSION

TO ensure that temperature values ob-

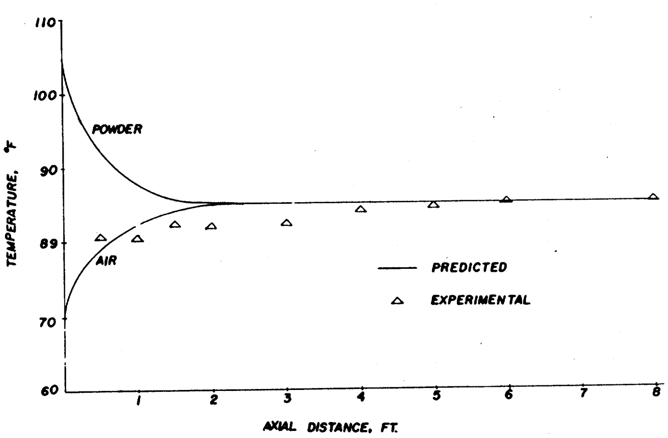

Fig. 3—Comparison of predicted and experimental temperatures during equilibrium with adiabatic conditions at wall.

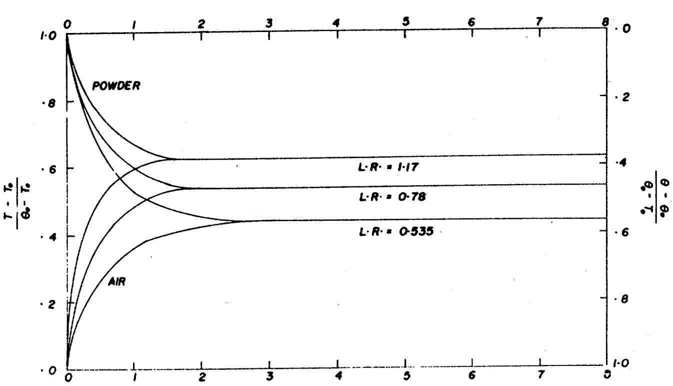

Fig. 4—Influence of product loading ratio on equilibration temperature and distance required to reach equilibration.

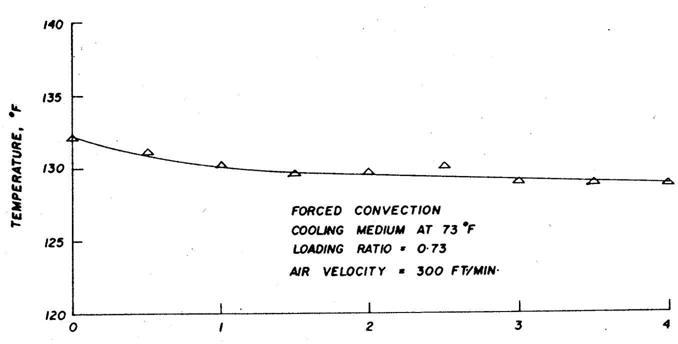

Fig. 5—Effectiveness of forced-air circulation on cooling product-air mixture after equilibration.

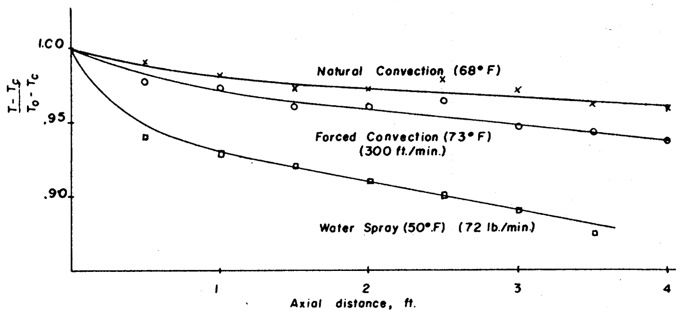

Fig. 6—Comparison of various conditions at the cooling surface on cooling effectiveness.

tained from measurements in the air-product mixture represented a mean value for conditions in the tube, the temperature profile was measured. These measurements were accomplished by using 4 thermocouples, equally spaced in the 2-in.-diameter tube. Since the product particles were concentrated in the lower portion of the tube, it was reasonable to expect (Wen and Simons, 1959) the mean of the temperature distribution to be below the tube center. Measurements of temperature distribution indicated that a thermocouple located 0.75 in. from the bottom of tube would provide an acceptable prediction of the mean temperature.

Temperature equilibration region

The solid curves in Figure 3 indicate agreement between predictions from Equations [12] and [14] and experimental data obtained by the described methods. These results indicate the agreement for data collected at a loading ratio of 0.535 with each experimental point representing a mean of at least 10 temperature values. The initial product temperature and initial air temperature were 105 and 70°F, respectively, and came to equilibrium at approximately 85°F.

The experimental data tend to be lower than the predicted curves and longer distance is required to reach the predicted equilibration temperature. This is probably related to limitations of the prediction equations. The possibilities that relative velocity between particle and air is not constant and that the product powder is not well dispersed during early portions of the equilibration region would contribute to the type of experimental results obtained.

Figure 4 illustrates the influence of different loading ratios. Equations [15] and [16] were used as the prediction equations in generating the curves. As the loading ratio increased, the equilibrium temperature increased. This is as expected, since the larger amount of warm product in the tube will increase the product-air mixture temperature.

Interestingly, equilibration distance decreases with an increase in loading ratio (Fig. 4). Physically, this means that as larger amounts of product are released into the air flowing in the tube, shorter distance is required for the product and air to reach some equilibration temperature. The equilibration occurs at temperatures closer to the product temperature and, in the limit, equilibration would occur instantaneously at the initial product temperature for very high loading ratios. At very low loading ratios, equilibration would require infinitely long distances at temperatures near the initial air temperature. The significance of equilibration temperature is that the extent of cooling during this portion of conveying is known and can be utilized in the cooling system design.

Effectiveness of various cooling surfaces

By exposing the surface of the tube to various conditions, different rates of product cooling below the equilibration temperature can be achieved. Experiments were conducted to compare natural circulation of air, forced circulation of air and spraying of cool water over the exterior surface of the conveying tube with a loading ratio of 0.73 lb product per lb air.

The results (Fig. 5) illustrate the extent of product-air mixture cooling which occurs in a 4-ft distance. The cooling medium (air) was 73°F and was flowing at a free stream velocity at 300 ft/min. Each experimental point represents the mean of 10 temperatures recorded during the experiment.

To compare the results of the three cooling methods, a dimensionless temperature ratio $\frac{T - T_c}{T_o - T_c}$ was used. Results in Figure 6 illustrate the effectiveness of the three methods. The water spray provides the most effective cooling of the product-air mixture, while forced-circulation air flow over the conveying tube is more effective than natural circulation. In the 4-ft distance investigated, water spray reduced the product-air mixture temperature by 12.5% of the initial temperature difference between the mixture and cooling medium. Force circulation and natural circulation reduced temperature by about 6 and 4%, respectively.

CONCLUSIONS

THE PRODUCT and air temperatures which exist during the equilibration region of pneumatic flow can be predicted with reasonable accuracy using equations derived from enthalpy balance.

An increase in loading ratio increases the equilibration temperature of the product-air mixture and reduces the distance required to reach equilibration.

Conditions at the conveying tube exterior surface influence cooling effectiveness significantly. A water spray over the exterior surface is the most effective of methods investigated, while forced air circulation is more effective than natural circulation of air.

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Ms. received 7/28/69; revised 7/1/70; accepted 7/14/70.

Presented at the 29th Annual Meeting of the Institute of Food Technologists in Chicago.

Journal Article No. 4786, Michigan Agricultural Experiment Station.

This study was supported by the Agricultural Research Service, U.S. Department of Agriculture, under Grant No. 12-14-100-9144(73) administered by the Eastern Utilization Research and Development Division, Philadelphia, Pennsylvania 19118.

The authors ackowledge the assistance provided by Dr. A.W. Farrall during the early stages of planning for this investigation.